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Minimum Viscosity for Bearing Reliability in R290 Rotary Compressors

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ABSTRACT

R290 which has been considered for use in room air conditioning systems as low global warming potential (GWP) refrigerant has attracted great attention. This refrigerant is determined the lubrication performance of the sliding parts (especially the bearings) in rotary compressor. It is important to prevent the sliding parts from seizure and wear. To ensure the bearing reliability, it is traditional for R290 compressor manufacturer to specify a minimum oil sump temperature difference above the saturated discharge temperature. It is a surrogate measure of minimum viscosity required to ensure adequate lubricant film thickness in bearings under the system flooding conditions. The minimum viscosity depends on the bearing geometry, the load and the speed. This paper describes a method for specifying the minimum safe viscosity of the high-side R290 rotary compressors based on lubricant film thickness calculations.

1. INTRODUCTION

As alternative refrigerant of R22, R410A doesn't destroy the atmosphere ozone layer, but it has a serious problem that its influence of green house effect is greater. As a kind of hydrocarbon refrigerants, R290 (propane) with nearly zero ozone depletion and global warming potential have been used as working fluid in room air conditioners. Although the drawback of R290 is flammability, it can be improved by optimized design in refrigeration systems, such as reducing the charged amount of refrigerant so that the risk of inflammation and explosion is reduced to the least.

And the exploitations of R290 compressors are in progress. One of the most important tasks for these exploitations is keeping the reliability of compressors. Lubrication is crucial to good reliability and high performance in a rotary compressor, as in other types of hermetic refrigeration compressors (Katsumi *et al*., 1996). On selecting the proper lubricating oil, the viscosity grade (VG) and the miscibility level with the refrigerant are important properties. As the lubricating oil, mineral oil (MO) has been most applicable for R22; synthetic oils (POE and PAG) are suit for R410A. These oils also have good miscibility with R290. However, good selection is hardly possible with only those static property data. Because under a dynamic situation such as inside the compressor during system operation, many factors may be different from those for the current R22/MO, such as the temperature distribution or ambient pressure, so the refrigerant and oil behavior can not always be the same as that for the current R22/MO (Takashi *and* Makoto, 1996).

The rotary compressor bearing is designed as a journal bearing to be lubricated by the condition, which depends on both the oil viscosity behavior and the load influenced by pressure and rotating frequency. It is typical that maximum load flood-back condition is part of the qualification to represent the maximum bearing load and minimum oil viscosity that will result in the minimum bearing film thickness of any continuous qualification test (Paul *et al*., 1998) (Akihiko *et al*., 1998) (Mitsuhiro *et al*., 1996) (Katsuya *et al*., 1998).

This paper describes a method for specifying the minimum safe viscosity of the high-side R290 rotary compressors based on lubricant film thickness calculations.

2. GOVERNING EQUATIONS

Fig.1 shows the cross section of the rotary compressor. The rotary compressor is a high side compressor with the lubricant subjected to discharge conditions. The bearings are the hydrodynamic journal bearings on the upper and lower sides and eccentric bearings. The schematic drawing of the compression part in a rotary compressor is shown in Figure 2. And Fig. 3 shows the schematic drawing of roller/crank and Fig. 4 shows its coordinate system for the analysis. The roller is loaded large dynamic forces such as gas forces, vane forces and unbalanced forces. Therefore, the Reynolds equations are used to calculate the reaction forces of oil film.

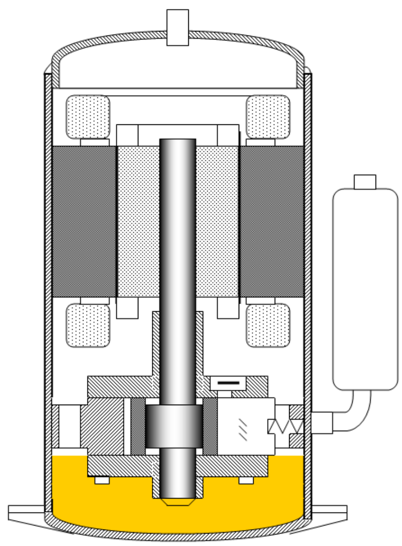

Figure 1: The cross section of the rotary compressor

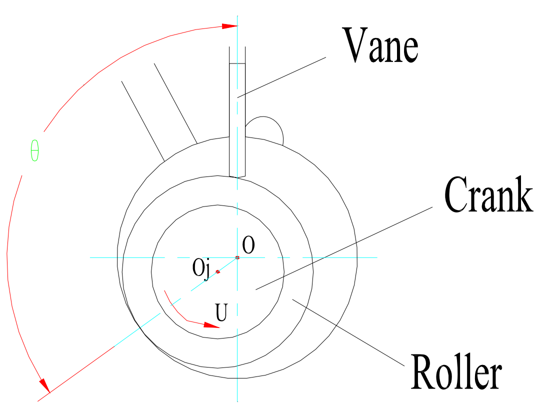

Figure 2: The schematic drawing of the compression part in a rotary compressor

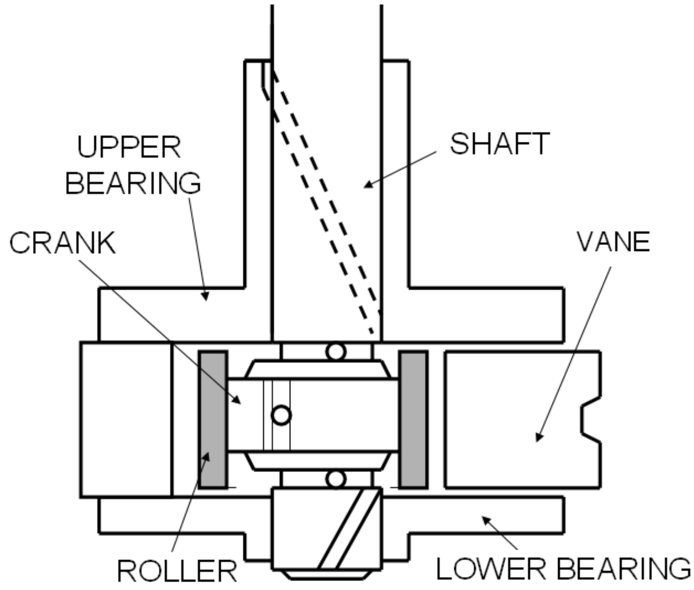

Figure 3: The schematic drawing of roller/crank

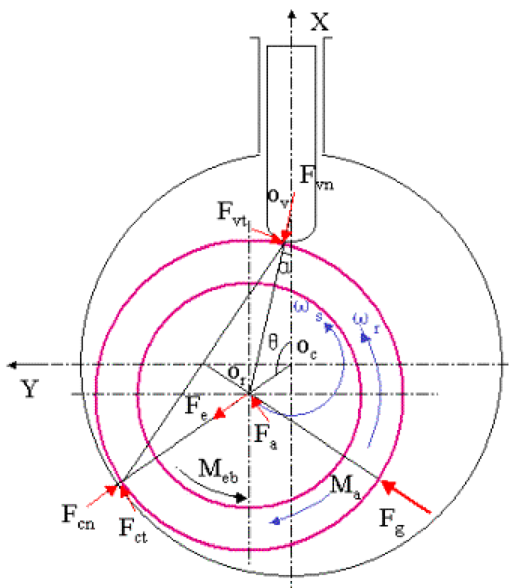

Figure 4: The roller/crank coordinate system for the analysis

2.1 Momentum Equations of the Roller

Momentum equations of the roller are simply expressed as following (Katsuya et al., 1998).

$$M \ddot{x} = Fx + Fcx \tag{1}$$

$$M \ddot{y} = Fy + Fcy \tag{2}$$

where m is the mass of the roller, Fx and Fy are the outer loads acting on the roller, Fcx and Fcy are the reaction forces of oil film.

2.2 Outer Loads

The outer loads acting on the roller are defined as following (Katsuya et al., 1998).

$$Fx = Fgx + Fvx + Fux \tag{3}$$

$$Fy = Fgy + Fvy + Fuy \tag{4}$$

Where Fgx and Fgy are the gas forces, Fux and Fuy are the unbalanced forces, Fvx and Fvy are the forces at the contact point with the vane. Fvx and Fvy are defined as following (Katsuya et al., 1998).

$$Fvx = Fsx + Fpx + Ffx \tag{5}$$

$$Fy = Fgy + Fvy + Fuy \tag{6}$$

Where Fsx and Fsy are the vane spring forces, Fpx and Fpy are the forces produced by the difference of gas pressures around the vane, Ffx and Ffy are the frictional forces between the vane and the vane slot.

2.3 Reaction Forces of Oil Film

The reaction forces of oil film between the roller and the crank are calculated by the Reynolds equations based on the short bearing theory as given by the following equations (Katsuya et al., 1998).

$$\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right) = 12\frac{U}{R}\frac{\partial h}{\partial \theta} + 12\frac{\partial h}{\partial t} \tag{7}$$

$$h = c - x_r \cos\theta - y_r \sin\theta \tag{8}$$

Where p is the oil pressure between the roller and the crank, $\mu$is the oil viscosity, U is the sliding speed, R is the radius of the crank, and c is the radial clearance. The reaction forces Fcx, Fcy are calculated by integrating the oil pressure which is obtained by equation (7) and (8).

2.4 Dynamic Bearing Load

The dynamic bearing loads of the upper bearing and lower bearing requires numerical solution to the Reynolds equations based on the finite length journal bearing theory as given by the following equations (Fei et al., 2006).

$$\frac{1}{R^2}\frac{\partial}{\partial \theta}\left(\frac{h^3}{\mu}\frac{\partial P_b}{\partial \theta}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial P_b}{\partial z}\right) = 6\frac{U}{R}\frac{\partial h}{\partial \theta} + 12\frac{\partial h}{\partial t} \tag{9}$$

$$\begin{Bmatrix} F_{bx} \\ F_{by} \end{Bmatrix} + R\int_0^L \int_{\theta_1}^{\theta_2} P_b \begin{Bmatrix} \cos\theta \\ \sin\theta \end{Bmatrix} d\theta dz = 0 \tag{10}$$

Where Pb is the oil pressure between the bearing (the upper bearing or lower bearing) and the shaft. It also means the bearing pressure. And the bearing forces Fbx, Fby are calculated by equation (10).

3. LUBRICANT FILM THICKNESS CALCULATIONS

3.1 Minimum Film Thickness Calculations

The bearing lubricant film thickness requires numerical solution to the Reynolds equation. For a bearing of a given diameter to length ratio λ (λ=D/L) , the minimum film thickness (Hmin) can be computed as a function of the Sommerfeld number to allow other similar bearing conditions to be determined. The Sommerfeld number S is given by the following equations (McGraw, 1995).

$$S = \left(\frac{D}{c}\right)^2 * \frac{\mathrm{N}^* \ \mu}{\mathrm{P_b}} \tag{11}$$

This includes all the variables from the designer's standpoint and is a dimensionless number. D, c, and N are constant in a given fixed-speed design. And the active variables are μ and P. The lubricant viscosity, μ, is that inside the bearing gap. However, we will employ the lubricant viscosity in the oil sump as a surrogate. It is determined based on the temperature and pressure of the refrigerant over the oil- refrigerant mixture in the sump. The minimum film thickness was computed for the bearings and it is shown in Fig. 5 (McGraw, 1995). The minimum film thickness variable is defined as a Hmin/c. So the minimum film thickness is as below.

$$H_{\min} = c * \frac{H_{\min}}{c} \tag{12}$$

3.2 Lubricant/R290 Mixture Viscosity Requirement

The minimum film thickness also directly affects the conventional lubrication regime determination using the parameter "Λ" (equation 13). Λ is typically used to determine the lubrication regimes that reasonably well demonstrate the contacting surfaces, which directly influences the wear behavior of a particular lubricated system. Typically, the lubrication regimes, by using theΛ parameter, are used as follows. Λ≥3 for hydrodynamic lubrication or elastohydrodynamic lubrication, 3≥Λ≥1 for mixed lubrication and 1≥Λ for boundary lubrication regime, which is schematically presented in Fig. 6. a more precise classification also includes Λ≥4 for fully elastohydrodynamic lubrication with no effects on wear, and mixed lubrication regime is also divided up from 1 to 1.5 and from 1.5 to 3, indicating the different influences on wear and lifetime (Kalin and Velkavrh, 2009) (Stachowiak and Batchelor, 2005) (Tallian, 1967).

$$\Lambda = \frac{H_{\min}}{\sqrt{R_1^{\ 2} + R_2^{\ 2}}} \tag{13}$$

In order to prevent bearing wear, the value of Λparameter must be larger than 3 under all operating conditions. The minimum film thickness can be estimated for any given sump temperature (Ts), suction pressure (Ps) and discharge pressure (Pd) from the bearing load (Pb) and the viscosity of the lubricant/refrigerant mixture in the sump (μ). If the bearing load Pb is a function of Pd and Ps, then the Sommerfeld number will yield μ_{min} as a function of Pd and Ps.

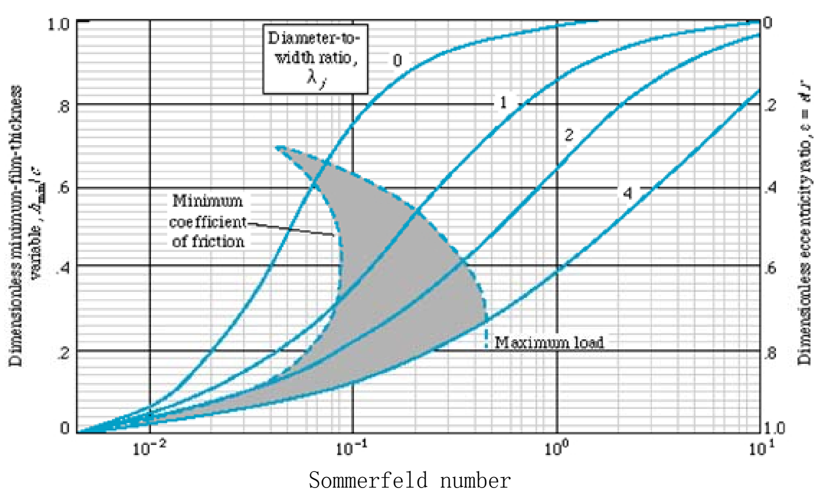

Figure 5: Minimum film thickness variable for bearings

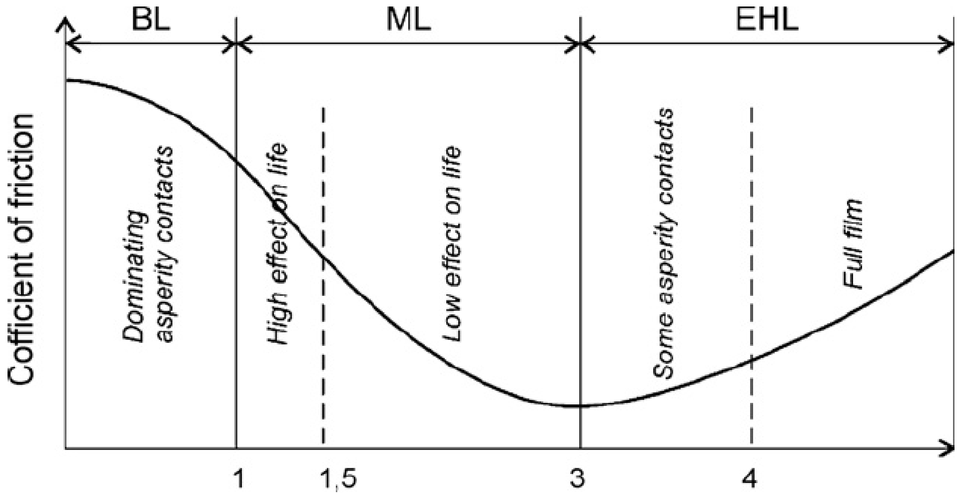

Figure 6: A schematic of a typical classification of different lubrication regimes depending on Λ parameter

3.3 Lubricant/R290 Mixture Properties

Fig.7 and Fig.8 show the lubricant/R290 mixture properties. The viscosity $\mu$is expressed as a function of pressure (P_{mix}) and solubility. And the solubility is a function of pressure (P_{mix}) and temperature (T_{mix}). Then the viscosity is a function of P_{mix} and T_{mix}. In a high side compressor where P_{mix}=Pd and T_{mix}=Ts, so the viscosity is a function of Pd and Ts. The dynamically loaded bearing film thickness can be represented in terms of the integrated average load on the bearing. Assuming dynamic similarity throughout the operating map of a given rotary compressor, the Sommerfeld number can be employed to provide a reasonable estimate of μ_{min} (Pd, Ps).

The lubricant/R290 mixture viscosity can be determined by the measurement of the pressure and temperature of the compressor sump. The sump pressure is very close to the discharge pressure. And the oil sump temperature can be measured easily. The lubricant/R290 mixture viscosity may be obtained from the diagrams as shown in Fig.7 and Fig.8. To bring the diagrams into perspective for a vapor compressor, a minimum mixture viscosity will be maintained by specifying a minimum temperature at discharge pressure. The discharge pressure and the mixture viscosity are the common variables that couple the load and lubricant/R290 mixture in the compressor. The bearing load is translated into a viscosity requirement through the Sommerfeld number.

$$P_b(Pd,Ps) \rightarrow \mathrm{S} \rightarrow \mu_{\min}(Pd,Ps) \tag{14}$$

The required viscosity at a certain operating condition only depends on the sump temperature (Ts). For a given compressor, the minimum allowable sump temperature (T_{min}) is only a function of Pd and Ps. So the minimum mixture viscosity can be maintained at any conditions.

4. CONCLUSIONS

This paper describes a method for specifying the minimum safe viscosity of the high-side R290 rotary compressors based on lubricant film thickness calculations. It is a surrogate measure of minimum viscosity required to ensure adequate lubricant film thickness in bearings under the system flooding conditions. The minimum lubricant/R290 mixture viscosity can be expressed as a function of discharge pressure and suction pressure. At any operating conditions, the compressor should be continuously operated with the mixture viscosity of the sump larger than the minimum mixture viscosity. From this minimum mixture viscosity, a minimum sump temperature will be specified. It can help the R290 compressor manufacturers to ensure the bearing reliability.

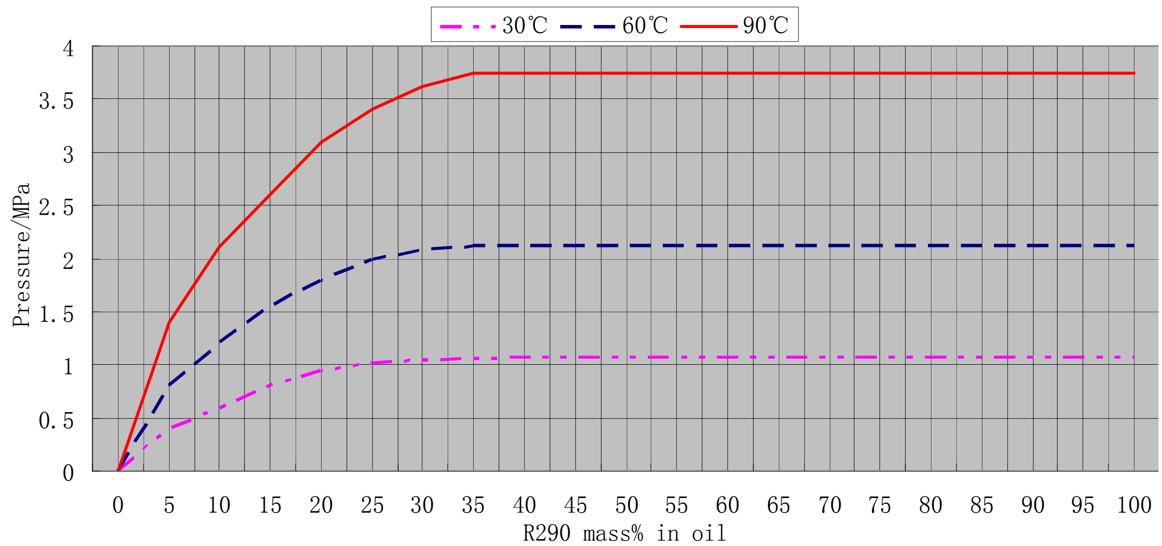

Figure 7: Solubility/pressure chart

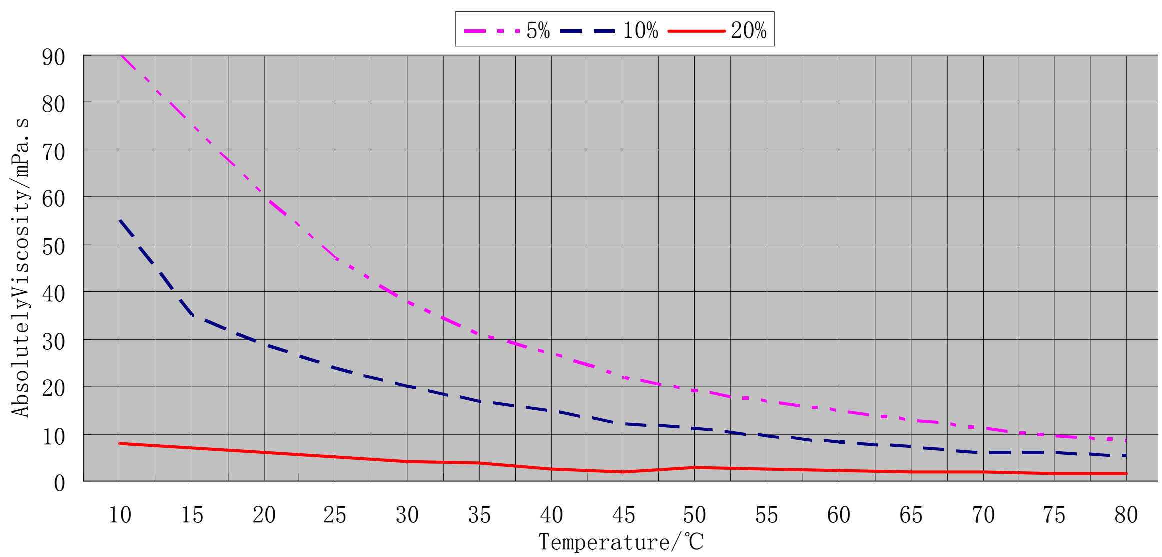

Figure 8: Viscosity /temperature chart

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